IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Application No. :

U.S. National Serial No. :

Filed :

PCT International Application No. : PCT/IB2004/003769

VERIFICATION OF A TRANSLATION

I, Charles Edward SITCH BA,

Deputy Managing Director of RWS Group Ltd UK Translation Division, of Europa House, Marsham Way, Gerrards Cross, Buckinghamshire, England declare:

That the translator responsible for the attached translation is knowledgeable in the French language in which the below identified international application was filed, and that, to the best of RWS Group Ltd knowledge and belief, the English translation of the international application No. PCT/IB2004/003769 is a true and complete translation of the above identified international application as filed.

I hereby declare that all the statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true; and further that these statements were made with the knowledge that willful false statements and the like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code and that such willful false statements may jeopardize the validity of the patent application issued thereon.

Date: March 30, 2006

Signature :

For and on behalf of RWS Group Ltd

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Vehicle air-conditioning assembly

The invention relates to motor vehicle air conditioning circuits.

In conventional motor vehicles, the compressor of the air conditioning circuit is driven by the engine and therefore consumes part of the engine's power. The power absorbed by the compressor, when it is operating, reduces the efficiency of the engine and consequently increases the fuel consumption and the pollution generated by the exhaust gases from the vehicle. This drawback is problematic in the case of mechanical compressors with an external control, the use of which is commonplace.

Moreover, in existing constructions, the fuel injection computer of the vehicle does not have the instantaneous value of the actual power absorbed by the compressor and therefore chooses, for the operation of the compressor, fuel injection parameters by default, which correspond to the maximum value of the power absorbed, which value is rarely reached in practice.

Consequently, one solution for optimizing the efficiency of the engine consists in estimating the instantaneous value of this power actually absorbed by the compressor. When this information is known, it is then possible to adapt the fuel injection parameters of the engine to the actual requirements.

In existing constructions, an estimate of the refrigerant mass flow rate is used to calculate the instantaneous power absorbed by the compressor.

Such constructions generally involving subcritical

refrigerants are unsuitable for supercritical refrigerants.

The use of supercritical refrigerants, especially the CO_2 refrigerant R744, was developed for vehicle air conditioning circuits in order to limit the deleterious effects of refrigerants on the environment. The CO_2 refrigerant has a much smaller global warming effect than subcritical refrigerants, such as the HFC refrigerants of the R134a type.

An air conditioning circuit using a supercritical fluid comprises a compressor, a gas cooler, an internal heat exchanger, an expander and an evaporator, the refrigerant passing through these in the above order. In such a circuit, the cooling of the refrigerant after compression undergoes no phase change. The refrigerant passes into the liquid state only during the expansion. This property of supercritical fluids means that the assembly given in patent application 01/16568 cannot be used to estimate the supercritical refrigerant flow rate and the power consumed by the compressor.

US 2003/0115896 A1 proposes an air conditioning unit for estimating the mass flow rate of a supercritical refrigerant, based on a measurement of the high pressure and a measurement of the low pressure. However, in order for the estimate for the mass flow rate to be sufficiently accurate, it is necessary for the air conditioning circuit to be controlled so that the refrigerant leaving the expander is almost entirely in the liquid state. Moreover, a sensor is required for measuring the low pressure, which increases the cost of the air conditioning unit.

French patent application 03/03362 also proposes an air conditioning unit for estimating the mass flow rate of

a supercritical refrigerant. To do this, the proposed air conditioning unit includes a calculating function that uses two differences in temperature relating to the gas cooler, at least one of which is based on the temperature of the refrigerant at a chosen intermediate point on the gas cooler. This intermediate point is in particular located at a distance x_i from the inlet of the gas cooler that is between 5% and 35% of the total length of the gas cooler. However, this assembly requires a large number of sensors (sensors for measuring the refrigerant pressure at the inlet and outlet of the compressor, the temperature of the refrigerant at the inlet of the gas cooler, the temperature of the stream of air received by the gas cooler and the temperature of the refrigerant at the chosen intermediate point on the gas cooler), which therefore increases the cost of the assembly.

The subject of the present invention is an air conditioning unit that remedies these known drawbacks of the prior art.

For this purpose, the invention proposes a motor vehicle air conditioning unit, provided with a supercritical refrigerant circuit comprising a compressor, a gas cooler, an expander, defining a refrigerant flow area, and an evaporator. The assembly further includes an electronic control device designed to interact with the refrigerant circuit. Advantageously, the electronic control device includes a calculating function using an estimate of the flow area of the expander, the density of the refrigerant and the pressure of the refrigerant at the inlet of the expander in order to calculate an estimate of the refrigerant mass flow rate at the expander.

According to one aspect of the invention, the flow area

of the expander is estimated from the refrigerant pressure at the inlet of the expander.

In particular, the electronic control device may be capable of reacting to the fact that the value of the refrigerant pressure P_{20} at the inlet of the expander is:

. less than or equal to a first pressure value P1, a first constant S1 being assigned to the flow area of the expander;

. less than or equal to a second pressure value P2 greater than the first pressure value P1, by solving the following equation in order to calculate an estimate of the flow area S of the expander:

$$S = S1 + (S2-S1) \times (P_{20}-P1)/(P2-P1),$$

where S2 is a second constant;

. less than or equal to a third pressure value P3 and greater than the second pressure value P2, solving the following equation in order to calculate an estimate of the flow area S of the expander:

$$S = S2 + (S3-S2) \times (P_{20}-P2)/(P3-P2),$$

where S3 is a third constant; and

. greater than or equal to the third pressure value P3, a fourth constant S4 being assigned to the flow area of the expander.

In one particular embodiment, the first pressure value P1 is approximately equal to 80 bar, the second pressure value P2 is approximately equal to 110 bar and the third pressure value P3 is approximately equal to 135 bar, while the first constant S1 is approximately equal to 0.07 mm^2, the second constant S2 is approximately equal to 0.5 mm^2, the third constant S3 is approximately equal to 0.78 mm^2 and the fourth constant S4 is approximately equal to 3.14 mm^2.

According to another aspect of the invention, the

calculating function is specific to calculating the density of the refrigerant from the refrigerant temperature at the inlet of the expander and from the refrigerant pressure at the inlet of the expander.

The air conditioning unit may include a probe placed at the inlet of the expander for measuring the refrigerant temperature at the inlet of the expander.

The air conditioning unit may also include a sensor placed at the inlet of the expander for measuring the refrigerant pressure at the inlet of the expander.

As a complement, the electronic control device may include a power estimation function capable of estimating the power absorbed by the compressor from:

- the refrigerant mass flow rate provided by the calculating function;
- the work of the compressor; and
- the rotation speed of the compressor.

The electronic control device is capable of estimating the work of the compressor from the refrigerant pressure at the inlet of the expander, from the refrigerant pressure at the inlet of the compressor and from a refrigerant temperature relative to the compressor.

Advantageously, the refrigerant pressure at the inlet of the compressor is estimated from a pressure at the inlet or at the outlet of the evaporator combined with the refrigerant mass flow rate.

Furthermore, the pressure at the inlet or at the outlet of the evaporator is determined from the refrigerant temperature, said temperature being either measured by a probe or estimated from:

- a temperature relative to the evaporator;
- the efficiency of the evaporator; and
- the temperature of the air to be cooled.

The refrigerant temperature relative to the compressor may be the refrigerant temperature at the inlet of the compressor.

The air conditioning unit may then include a probe placed at the inlet of the compressor for measuring the refrigerant temperature at the inlet of the compressor.

As a variant, the refrigerant temperature relative to the compressor may be the refrigerant temperature at the outlet of the compressor.

The air conditioning installation may then include a probe placed at the outlet of the compressor for measuring the refrigerant temperature at the outlet of the compressor.

The invention also covers a product-program, which may be defined as comprising the functions used for estimating the refrigerant mass flow rate and the power consumed by the compressor.

Other features and advantages of the invention will become apparent on examining the detailed description below and the appended drawings in which:

. Figure 1A is a diagram of a motor vehicle air conditioning circuit operating with a supercritical refrigerant;

. Figure 1B is a diagram of an air conditioning unit according to the invention;

. Figure 2 is a plot showing the variations in the flow area of the expander as a function of the refrigerant pressure at the inlet of the expander;

. Figure 3 is a flow diagram showing the steps implemented by the control device for estimating the refrigerant mass flow rate at the expander;

. Figure 4 is a flow diagram showing the steps implemented by the control device for estimating the flow area of the expander;

. Figure 5 is a flow diagram representing the steps implemented by the control device for estimating the power consumed by the compressor, according to the invention;

. Figures 6 and 7 are diagrams of the air conditioning circuit according to alternative embodiments of the invention;

. Figures 8 to 12 are schematic diagrams showing the position of the temperature probes used for determining the pressure at the inlet or at the outlet of the evaporator;

. Figure 13 illustrates the method of determining the pressure of the refrigerant at the inlet of the compressor; and

. Figure 14 shows the relationship between the refrigerant mass flow rate and the refrigerant pressure at the inlet of the compressor.

Appendix A comprises the main mathematical equations used for implementing the assembly.

The drawings essentially contain elements having a certain character. They therefore serve not only for making the description more clearly understood but also for contributing to the definition of the invention, where appropriate.

Figure 1A represents an air conditioning circuit through which a supercritical refrigerant flows. Hereafter, the description will refer to the supercritical refrigerant CO_2 as a nonlimiting example.

Such a circuit conventionally comprises:

. a compressor 14 suitable for receiving the refrigerant in the gaseous state and compressing it;

. a gas cooler 11 suitable for cooling the gas compressed by the compressor;

. an expander 12 suitable for lowering the pressure of the refrigerant; and

. an evaporator 13 suitable for making the refrigerant from the expander pass from the liquid state to the gaseous state in order to produce a stream of conditioned air 21 that is sent into the passenger compartment of the vehicle.

The circuit may further include an internal heat exchanger 23, allowing the refrigerant flowing from the gas cooler to the expander to give up heat to the refrigerant flowing from the evaporator to the compressor. The circuit may further include an accumulator 17 placed between the outlet of the evaporator and the inlet of the compressor, in order to avoid liquid surges.

The gas cooler 11 receives a stream of external air 16 for extracting the heat taken from the passenger compartment, which stream under certain operating conditions is blown by a motor/fan unit 15.

The evaporator 13 receives a stream of air from a blower, in order to produce a stream of conditioned air 21.

The expander 12 may have a variable flow area, such as an electronic expander, a thermostatic expander or any other expander for which the flow area depends on the high pressure. The expander 12 may also have a fixed flow area, such as a calibrated orifice.

The supercritical refrigerant is compressed in the gaseous phase and raised to a high pressure by the compressor 14. The gas cooler 11 then cools the refrigerant by means of the incoming stream of air 16. Unlike the air conditioning circuits operating with a subcritical refrigerant, the cooling of the refrigerant after compression does not involve a phase change. The refrigerant passes to the two-phase state, with a vapor content that depends on the low pressure, only during the expansion. The internal heat exchanger 23 allows the refrigerant to be very strongly cooled.

Referring now to Figure 1B, this shows an air conditioning unit according to the invention installed in a motor vehicle.

The motor vehicle is driven by an engine 43, which may be controlled by a fuel injection computer 42. The fuel injection computer 42 receives information from various sensors, which it interprets in order to adjust the injection parameters.

The fuel injection computer 42 may also provide information about the conditions inside or outside the vehicle (information provided by a solar sensor, number of occupants, etc.). In particular, it may provide information about instantaneous values relating to the operation of the vehicle, and especially about the rotational speed N of the compressor.

The unit is also provided with an air conditioning computer 40, comprising a passenger compartment regulator 41 and an air conditioning loop regulator 402. The passenger compartment regulator 41 is designed to set the temperature setpoint of the external air blown into the evaporator 13.

The engine fuel injection computer may act on the air conditioning unit via an air conditioning regulator 402. This link may prevent the operation of the air conditioning unit when the engine is on high load.

The air conditioning unit according to the invention is based on a model of the expander, in order to provide an estimate of the refrigerant mass flow rate m_{exp} at the expander.

The air conditioning unit includes an electronic control device, for example an electronic card 401, designed to interact with the air conditioning circuit 10 via the links 30/31 and with the fuel injection computer 42 via the links 32/33.

The electronic card 401 may be considered as an integral part of the vehicle air conditioning computer 40.

The electronic card 401 may receive information 30 coming from sensors fitted on the air conditioning circuit 10. It may also receive information from the engine fuel injection computer 42 via the link 33, in particular the rotation speed N of the compressor and/or the run speed V of the vehicle.

The Applicant has found that the expander may be modelled by equation A10 of Appendix A, where K is a coefficient that characterizes the expander, in particular its pressure drop.

From this model, the calculating function can calculate an estimate of the refrigerant mass flow rate m_{exp} at the expander from:

. the pressure P_{20} of the refrigerant at the inlet

of the expander;

- the density ρ of the CO_2 refrigerant; and
- the flow area S (in mm^2) of the expander.

The Applicant has also found that the density ρ of the CO_2 refrigerant may be estimated from the temperature T_{30} at the inlet of the expander and from the pressure P_{20} at the inlet of the expander, according to equation A11 of Appendix A.

The Applicant has furthermore found that the flow area S (in mm^2) of an expander with a flow area depends on the refrigerant pressure P_{20} at the inlet of the expander.

Thus, an estimate of the refrigerant mass flow rate m_{exp} at the expander may be obtained from the refrigerant pressure P_{20} at the inlet of the expander and from the temperature T_{30} at the inlet of the expander.

The refrigerant pressure P_{20} at the inlet of the expander and the refrigerant temperature T_{30} at the inlet of the expander may be estimated or measured.

The air conditioning circuit may include two separate sensors for measuring the refrigerant pressure P_{20} at the inlet of the expander and the refrigerant temperature T_{30} at the inlet of the expander, respectively. As a variant, the air conditioning circuit may have a single sensor placed at the inlet of the expander for measuring both these quantities.

Figure 2 is a plot showing the variation in the flow area S (in mm^2) as a function of the refrigerant pressure P_{20} (in bar) at the inlet of the expander. The curves shown on this plot correspond to equations A2 to A5 of Appendix A.

As long as the refrigerant pressure P_{20} at the inlet of the expander is less than or equal to a first pressure value P1, the area S is equal to a first constant S1 according to equation A2 of Appendix A.

When the pressure P_{20} is above the first pressure value P1 and less than or equal to a second pressure value P2, the area S follows a straight line whose directrix is determined from the values of S1, P1, P2 and a second constant S2, in accordance with equation A3 of Appendix A. The value of S2 corresponds to the value of the area S when P_{20} is equal to the value P2.

When the pressure P_{20} is greater than the second pressure value P2 and less than or equal to a third pressure value P3, the area S follows a straight line whose directrix is determined from the values of S2, P2, P3 and a third constant S3 in accordance with equation A4 of Appendix A.

When the pressure P_{20} is greater than or equal to the third pressure value P3, the area S is equal to a fourth constant S4 greater than the third constant S3 in accordance with equation A5 of Appendix A.

In particular, the first pressure value P1 may be approximately equal to 80 bar, the second pressure value P2 may be approximately equal to 110 bar, the third pressure value P3 may be approximately equal to 135 bar, the first constant S1 may be approximately equal to 0.07 mm^2, the second constant S2 may be approximately equal to 0.5 mm^2, the third constant S3 may be approximately equal to 0.78 mm^2 and the fourth constant S4 may be approximately equal to 3.14 mm^2.

As a complement, the estimate of the refrigerant mass

flow rate, provided by the calculating function of the electronic card, may be used to calculate the mechanical power absorbed. To do this, the electronic card includes a power estimation function capable of estimating the power P_{abs} absorbed by the compressor from the refrigerant mass flow rate m_{exp}. In particular, the power estimation function is capable of estimating the power P_{abs} absorbed by the compressor from the isentropic work of compression W_{ise} and from the rotation speed N of the compressor in accordance with equation A6 of Appendix A. The coefficients a and b are related to operating parameters of the air conditioning circuit. The coefficient a corresponds to the mechanical efficiency relative to the isentropic compression of the compressor and is around 1.38. The coefficient b is the image of the compressor efficiency and corresponds to the friction factor of the compressor.

In accordance with equation A7 of Appendix A, the isentropic compression power W_{ise} is related:

. to the refrigerant mass flow rate m_{exp}, an estimate of which is calculated by the calculating function as described above; and

. to the isentropic work ΔH_{ise} of the compressor.

The Applicant has found that the estimate of the work ΔH_{ise} of the compressor may be obtained, in accordance with equation A80 of Appendix A, from:

. the refrigerant pressure P_{20} at the inlet of the expander;

. the refrigerant pressure P_{35} at the inlet of the compressor; and

. the temperature of the refrigerant T_{comp} relative to the compressor.

The refrigerant pressure P_{35} at the inlet of the

compressor and the temperature of the refrigerant T_{comp} relative to the compressor may be estimated or measured.

The pressure of the refrigerant at the inlet of the compressor is estimated using the refrigerant mass flow rate m_{exp} calculated above and from the pressure drop Δp between the inlet of the evaporator 13 and the inlet of the compressor 14.

As a variant, this estimate of the refrigerant pressure at the inlet of the compressor may be determined using the refrigerant mass flow rate m_{exp} calculated above and from the pressure drop Δp between the outlet of the evaporator 13 and the inlet of the compressor 14.

The example below is described in relation to the inlet pressure of the evaporator 13, but this example can be transposed in a similar manner using the pressure at the outlet of the evaporator 13.

This pressure drop Δp is calculated from formula A90 of Appendix A, in which:

- P_{50} is an estimate of the pressure at the inlet of the evaporator;
- P_{35} is the estimate of the refrigerant pressure at the inlet of the compressor.

It is also known that this pressure drop Δp may be determined from formula A100 in which:

- K is a pressure drop coefficient;
- ρ is the density of the refrigerant; and
- V_{co_2} is the speed of the refrigerant.

In accordance with equation A101 of Appendix A, the speed of the refrigerant V_{co_2} can be determined from:

- the refrigerant mass flow rate m_{exp} determined by means of equation A10;

- the refrigerant density ρ; and

- a constant S corresponding to the mean flow area and mean flow length, combining both the linear pressure drop and the singular pressure drop, experienced by the refrigerant. From the two equations A90 and A100 it is possible to estimate the pressure of the refrigerant at the inlet of the compressor, as illustrated by equation A9.

Thus, the only unknown in this equation is the estimate of the pressure P_{50} at the inlet of the evaporator.

It is known from the fluids saturation law, otherwise known as the fluid state law, that the pressure P_{50} at the inlet of the evaporator depends directly on the saturation temperature T_{50} at the inlet of the evaporator. Over the operating range of interest to us, this equation may be represented in the form of a second-order polynomial.

This saturation temperature T_{50} of the pressure refrigerant may be measured or estimated.

When it is estimated, equation A91 of Appendix A is used, in which:

- T_{40} is a datum relating to the evaporator temperature available on many air conditioning units. This involves a CTN or CTP probe placed on the evaporator, the main objective of which in the prior art is to prevent the evaporator from icing up when the compressor stops. This temperature T_{40} corresponds to the surface temperature of one of the walls of the evaporator 13 (for example at the recess of the inserts, as illustrated in Figure 12) or to the air temperature at the outlet of the evaporator;

- η_{evap} is the image of the evaporator efficiency, which can be easily related to a function of the

voltage U for the air blower of the evaporator in the passenger compartment and the run speed V of the vehicle, as expressed in equation A910 of Appendix A; and

- T_{60} is the temperature of the air to be cooled by the air conditioning unit. This temperature is estimated according to the temperature inside the passenger compartment, the temperature outside the passenger compartment, the voltage U for the blower, the position of the recycling flap of the air conditioning unit and the run speed V of the vehicle. This function is expressed in equation A920 of Appendix A.

Figures 8 and 9 illustrate the possibility of measuring the saturation temperature T_{50} of the refrigerant at the inlet of the evaporator 13 either by means of an intrusive or direct temperature probe 51, that is to say a probe directly bathed by the refrigerant (Figure 7), or by a nonintrusive or indirect probe 52, which measures the temperature of the refrigerant on the basis of the temperature of the tube that is transporting it (Figure 8).

Figures 10 and 11 show the possibility of measuring the saturation temperature T_{50} of the refrigerant at the outlet of the evaporator 13 using means identical to the temperature measurement envisaged at the inlet of the evaporator 13.

This method of determining the refrigerant pressure at the inlet of the compressor is summarized in Figure 13 in the form of a block diagram in which:

- if the temperature T_{50} at the inlet or at the outlet of the evaporator is estimated, then the following are used:
  - o the evaporator efficiency η_{evap} determined from

the information available about the vehicle, such as blower voltage U and the run speed V of the vehicle,

o these two items of information are used also to determine the temperature T60 of the air to be cooled, combined with the temperature inside the passenger compartment and the external temperature and

o the evaporator efficiency η_{evap}, the temperature T_{60} of the air to be cooled and the surface temperature T_{40} of the evaporator are combined to estimate the temperature T_{50} at the inlet of the evaporator 13;

- if the temperature T_{50} is measured, a temperature probe 51 or 52 delivers the expected value;

- the estimated or measured temperature T_{50} is used to determine the pressure P_{50} according to the refrigerant saturation law;

- this pressure P_{50} at the inlet or the outlet of the evaporator 13, combined with the refrigerant pressure P_{20} at the inlet of the expander, with the density ρ of the CO_2 refrigerant and the flow area S (in mm^2) of the expander allows the refrigerant mass flow rate to be determined; and

- finally, the combination of this mass flow rate information with the estimate of the pressure P_{50} at the inlet of the evaporator allows the refrigerant pressure P_{35} at the inlet of the compressor to be determined without using a specific sensor and thus without increasing the cost of the air conditioning unit.

Figure 14 illustrates the relationship between the refrigerant mass flow rate and the refrigerant pressure P_{35} at the inlet of the compressor. The x-axis of this curve represents the mass flow rate m_{exp} in kilograms per hour and the y-axis of this curve illustrates the pressure drop Δp in bar between the inlet or the outlet of the evaporator 13 and the inlet of the compressor 14. It may be seen that an error of around 30 kg in the

estimate of the mass flow rate results in an error of around 2 bar in the determination of P_{35}. This error is minor compared with the absolute operating pressure values, which are often greater than 35 bar.

The refrigerant temperature relative to the compressor may be the refrigerant temperature T_{35} at the inlet of the compressor, in accordance with equation A81 of Appendix A. R is the perfect gas constant and M corresponds to the molar mass of the refrigerant. The ratio R/M may especially be equal to 188.7.

As a variant, the refrigerant temperature relative to the compressor may be the refrigerant temperature T_{36} at the outlet of the compressor in accordance with equation A82 of Appendix A.

Figure 3 is a flow diagram showing the steps implemented by the electronic card for estimating the refrigerant mass flow rate m_{exp} and the power consumed by the compressor.

At step 100, the refrigerant pressure P_{20} at the inlet of the expander is estimated/measured. Referring to Figures 1B, 6 and 7, the refrigerant pressure P_{20} at the inlet of the expander may be measured by a sensor 20 placed at the inlet of the expander. As a variant, the pressure P_{20} of the refrigerant at the inlet of the expander may be estimated.

At step 102, the electronic card 401 estimates the flow area S of the expander 12 from the measured/estimated refrigerant pressure value P_{20} at the inlet of the expander in accordance with equations A4 and A5 of Appendix A.

Step 102 is shown in detail in the flow diagram of

Figure 4. The electronic card determines if the measured value of the refrigerant pressure P_{20} at the inlet of the expander is:

- less than or equal to the first pressure value P1 (step 1020), in which case the flow area of the expander is equal to S1;
- greater than the first constant P1 and less than or equal to the second pressure value P2 (step 1021), in which case the flow area of the expander is given by equation A3 of Appendix A as a function of the pressure P_{20} obtained at step 100;
- greater than the second pressure value P2 and less than or equal to the third pressure value P3 (step 1022), in which case the flow area of the expander is given by equation A4 of Appendix A, as a function of the pressure P_{20} obtained at step 100; and
- greater than or equal to the third pressure value P3 (step 1023), in which case the flow area of the expander is equal to S4.

At step 103, the electronic card provides an estimate/measurement of the temperature T_{30} at the inlet of the expander. The unit may include a temperature sensor 30 for measuring the refrigerant temperature T_{30} at the inlet of the expander, as shown in Figures 1B, 6 and 7. As a variant, the unit may include a single sensor 20 for measuring both the pressure P_{20} and the temperature T_{30} of the refrigerant at the inlet of the expander. The refrigerant temperature T_{30} at the inlet of the expander may also be estimated.

At step 104, the electronic card 401 estimates the density ρ of the CO_2 refrigerant. The density ρ of the CO_2 refrigerant may be calculated according to equation A11 of Appendix A from the refrigerant pressure P_{20} at the inlet of the expander, obtained at step 100, and from the refrigerant temperature T_{30} at the inlet of the the

expander, obtained at step 103.

At step 105 of Figure 3, the electronic card 401 can then calculate the refrigerant mass flow rate m_{exp} according to equation A3 of Appendix A, from:

. the refrigerant pressure P_{20} at the inlet of the expander, estimated/measured at step 100;

. the flow area S (in mm^2) of the expander, estimated at step 102; and

. the density ρ of the CO_2 refrigerant, estimated at step 104.

As a complement, the estimate of the refrigerant mass flow rate m_{exp} at the expander may be used to calculate the power P_{abs} consumed by the compressor.

Figure 5 is a flow diagram showing the steps implemented by the electronic card for calculating the power P_{abs} consumed by the compressor from the estimate of the refrigerant mass flow rate m_{exp} at the expander. The estimate of the power absorbed by the compressor may in particular require a prior estimate of the work ΔH_{ise} of the compressor, in accordance with equations A6 and A7 of Appendix A.

At step 200, the electronic card calculates an estimate of the work ΔH_{ise} of the compressor in accordance with equation A8 of Appendix A from:

. the refrigerant pressure P_{20} at the inlet of the expander;

. the refrigerant pressure P_{35} at the inlet of the compressor; and

. the refrigerant temperature T_{comp} relative to the compressor.

When the refrigerant temperature relative to the compressor is the refrigerant temperature T_{35} at the

inlet of the compressor (in accordance with equation A81 of Appendix A), it can be measured by a probe 35 placed at the inlet of the compressor, as shown in Figures 1B and 6.

When the refrigerant temperature relative to the compressor is the refrigerant temperature T_{36} at the outlet of the compressor (in accordance with equation A82 of Appendix A), it can be measured by a probe 36 placed at the outlet of the compressor, as shown in Figure 7.

The refrigerant pressure P_{35} at the inlet of the compressor may be estimated or measured.

At step 202, the electronic card calculates the isentropic power W_{ise} from the refrigerant mass flow rate m_{exp} obtained at step 105 and the work ΔH_{ise} of the compressor obtained at step 200, in accordance with equation A7 of Appendix A.

At step 204, the electronic card calculates an estimate of the power P_{abs} absorbed by the compressor, according to equation A6 of Appendix A, from the isentropic power W_{ise} obtained at step 202 and the rotation speed N of the compressor.

The rotation speed N of the compressor is supplied to the electronic card by the engine fuel injection computer 42 via the link 33, with reference to Figure 1B.

The computer may use the estimated value of the actual power consumed by the compressor to adjust the injection parameters, thereby making it possible to reduce the fuel consumption.

The air conditioning unit according to the invention makes it possible to obtain a satisfactory estimate of the refrigerant mass flow rate at the expander. Furthermore, this unit does not use a low-pressure sensor to estimate the refrigerant mass flow rate at the expander, thereby allowing the total cost of the unit to be reduced.

Of course, the present invention is not limited to the embodiments described above. It encompasses all alternative embodiments that a person skilled in the art might envision.

The present invention also covers the software code that it uses, most particularly when this is available on any medium that can be read by a computer. The expression "medium that can be read by a computer" covers a storage medium, for example a magnetic or optical storage medium, and also a transmission means, such as a digital or analog signal.

APPENDIX A

Supercritical refrigerant mass flow rate

$m_{exp} = KS\rho \ln(P_{20} + C)$ **(A10)**

$\rho = f(T_{30},\ P_{20})$ **(A11)**

Flow area S of the expander

IF $P_{20} \leq$ P1:

$S = S1$ **(A2)**

If P1 < $P_{20} \leq$ P2:

$S = S1 + (S2 - S1)\ (P_{20} - P1)/(P2 - P1)$ **(A3)**

If P2 < $P_{20} \leq$ P3:

$S = S2 + (S3 - S2)/(P_{20} - P2)/(P3 - P2)$ **(A4)**

If P_{20} > P3:

$S = S4$ **(A5)**

Power consumed by the compressor

$P_{abs} = aW_{ise} + bN$ **(A6)**

Isentropic power W_{ise}

$W_{ise} = m_{exp}\Delta H_{ise}$ **(A7)**

Work ΔH_{ise} of the compressor

$\Delta H_{ise} = f(P_{20},\ P_{35},\ T_{35})$ **(A80)**

$\Delta H_{ise} = [(P_{20}/P_{35})^{(k-1)/k} - 1](T_{35} + 273.15)(R/M)/(k-1)$ **(A81)**

$AH_{ise} = [1-(P_{20}/P_{35})^{(1-k)/k}](T_{36} + 273.15)(R/M)/(k-1)$ **(A82)**

Estimate of the refrigerant pressure P_{35} at the inlet of the compressor

$\Delta p = P_{50} - P_{35}$ **(A90)**

$\Delta p = k\ \rho V Co_2 / 2$ **(A100)**

$V_{CO_2} = M_{cxp} / \ (\rho S)$ **(A101)**

$P_{35} = P_{50} - M_{exp} K\ (2\rho S^2)$ **(A9)**

$T_{50} = (\ T_{40} - (1-\eta_{evap})\ T_{60}) / \eta_{evap}$ **(A91)**

T_{50} => P_{50} R744 refrigerant saturation law